

**"STRUCTURAL MODELLING OF DARRIEUS ROTOR WIND TURBINES
WITH HINGED BLADES AND STABILIZING STRUTS"**

Final Report

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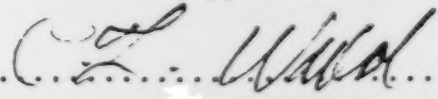
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ABSTRACT

This report presents the findings of a study of the structural mechanics of two-bladed Darrieus rotor wind turbines which are equipped with vertically-oriented hinges that allow the blades to move out of the plane of the rotor. The study examines, in particular, the structural dynamic behaviour of the Adecon 19-m rotor and a hypothetical Indal 6400 modified to incorporate hinged blades.

The report summarizes two earlier reports which were concerned with the Adecon 19-m rotor only. The report also presents full details of the natural frequencies, the frequency response (during normal operation), and the response to transients of the modified Indal 6400.

The conclusions of the study are that, while the inclusion of blade hinges may reduce some of the out-of-plane response during operation, the in-plane stresses are not attenuated. The result is a modest decrease in the maximum rms of cyclic stresses.

The advantages that accrue during normal operation may be offset by the difficulty of limiting peak stresses during braking. The flexible struts examined for the modified Indal 6400 must be restrained by other stiffer and stronger members during this transient condition.

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1.0 INTRODUCTION

1.1 Background

The aerodynamic principles of the Darrieus rotor have been well established. The performance of these wind turbines has been modelled accurately by the double multiple streamtube (1) and vortex shedding methods (2). The critical factor in establishing the economic viability of Darrieus rotor wind turbines is now the dynamic response of the structure (3) and the avoidance of fatigue failure (4,5). The ability to accurately predict the dynamic response during normal operation and during transient conditions has, therefore, become of prime importance.

Until recently the aerodynamic torque generated by these rotors has been transferred into the central column through a rigid connection at either end of the blade and also, on some machines, through horizontal struts connecting the column with a point on the blade. This design has led to bending stresses in the blade which tend to be critical factors in establishing the fatigue life of the rotor.

An alternative arrangement, which is being explored by Adecon Energy Systems Inc. (6), is to introduce, at upper and lower blade connections, a vertically orientated pin which is offset from the centre of the column. The effect of this during normal operation is to allow the blade to rotate about the vertical pin until the eccentricity of the axial tension in the blade is sufficient to transfer the mean torque into the central column.

The advantage of such a configuration is that the bending moments and stresses in the blade are much reduced. The fatigue life may be increased or, equivalently, lighter blade construction may be used without reducing fatigue life. Possible disadvantages of the configuration include excessive motion of the blade during starting and stopping and instability while the rotor is stationary unless this motion is controlled.

One arrangement which may overcome these disadvantages is the inclusion of flexible horizontal struts. These members would be in approximately the same location as the present horizontal struts on the Indal 6400 VAWT, but would comprise a single flexible member. It may be possible for these struts, if made from a fibre composite material, to combine the necessary flexibility for normal operation with the required strength for transient loading.

1.2 Methodology

The project was divided into six tasks:

- | | |
|---------|---|
| Task #1 | Natural Frequencies of Adecon 19-m |
| Task #2 | Frequency Response of Adecon 19-m |
| Task #3 | Natural Frequencies of Modified
Indal 6400 |
| Task #4 | Frequency Response of Modified Indal 6400 |
| Task #5 | Transient Response of Modified Indal 6400 |
| Task #6 | Final Report |

Results of the first two tasks have already been presented to the client but a summary of the results is

included in Section 2 of the present report. The remaining sections of this report are concerned with those tasks which pertain to the modified Indal 6400 (with hinged blades). Section 6 presents a summary and recommendations for the entire project and considers the results from examination of both the Adecon 19-m and the modified Indal 6400 rotors.

1.3 Statement of Work

Apart from summarizing the findings of the entire project, this report is concerned primarily with Tasks #3, 4 and 5. The Statement of Work included in the contract document for those tasks is therefore reproduced below.

1.3.1 Task #3. Natural Frequencies of Modified Indal 6400

Objectives

1. To modify the configuration of the present Indal 6400 to make the blades hinged at their roots and to identify a compatible flexible strut.
2. To investigate the natural frequencies of this modified rotor during normal operation.

Approach

1. A characteristic of the Adecon rotor is the low mass of the blade in contrast to which the blade of the Indal 6400 is very much heavier. This (and the larger overall dimensions) leads to higher

centrifugal forces in the blade of the Indal 6400, which in turn results in a lower blade offset required to power a rotor with hinged blades. If the eccentricity of the hinge is increased to 15% of the maximum radius, then calculations suggest that the blade offset can be limited to 8 degrees. The deflection required of any horizontal struts will be similarly limited.

2. Estimates will be made of a possible Indal 6400 modified to include hinged blades and flexible struts. Based on these estimates the basic finite element data will be prepared.
3. The techniques established in Task #1 for the determination of the natural frequencies will be applied to the modified Indal 6400.
4. The configuration may be further adjusted until an acceptable fan plot (natural frequencies vs. rotor speed) is obtained.

1.3.2 Task #4. Frequency Response of Modified Indal 6400

Objectives

1. To determine the response of the modified Indal 6400 as defined above under normal operation.
2. To make preliminary design of the flexible struts required for this rotor.

Approach

1. Due to the stiffness offered by the horizontal struts, there will be no zero eigenvalues associated with this (stationary) rotor and a procedure that uses the natural modes of the stationary rotor may be used. However, if the finite deformation effects are to be considered, then the same direct solution procedure outlined for Task #2 will be used.
2. The dynamic response of the operating rotor will be determined for a range of windspeeds. With this information the LIFE program can be used to estimate the fatigue life of the components that appear to be critical.
3. The design of the horizontal struts may require the use of a fibre composite material (wound glass fibre or higher strength fibre such as carbon or an aramid such as Kevlar). Some of the design specifications for the struts will be defined by this task.

1.3.3 Task #5. Transient Response of Modified Indal 6400

Objectives

1. To determine the response of the modified Indal 6400 under the action of survival winds.
2. To determine the response of the same configuration to braking loads.
3. To finalize the specifications for the flexible horizontal struts and to carry out preliminary designs.

Approach

1. There are two conditions that must be considered for the stationary rotor. One is the possibility of the upwind blade buckling under a survival wind condition. The second is the possible excitation of natural response under gusty wind conditions. The first condition can be solved by determining the steady-state aerodynamic loading and carrying out a NASTRAN Solution 64 (large displacement statics) in which the loading is applied incrementally. This has already been done to check the stability of earlier configurations.
2. The transient conditions during braking must consider both the rotating and the (nearly) stationary states. Critical conditions on a near-stationary rotor can be modelled through NASTRAN Solution 27 (direct transient response) as a step increase in the torque at the brake disc.
3. The results from these analyses will provide additional information on the structural integrity of the rotor, especially that of the flexible struts. A more complete set of specifications for the struts will be generated and the implications on their design will be discussed.

2.0 REVIEW OF TASKS #1 & 2 (Adecon 19-m)

2.1 Objectives

The objective of these tasks was to develop a NASTRAN-based procedure for determining the actual modes and frequencies of a Darrieus rotor with hinged blades and to apply the procedure to the Adecon 19-m rotor. A second objective was to examine the frequency response of that machine, attempt a prediction of fatigue life, and to compare with any field data available.

2.2 Finite Element Model of Adecon 19-m

The finite element idealization used to model the rotor is shown in Figure 2.1. The geometry and structural properties for the elements are based on details received from Adecon Energy Systems Inc. The stiffness of the upper and lower restraints are important and these were based upon a separate (static) analysis carried out on the external skeleton used to support this rotor.

The upper and lower supports are connected through the supporting skeleton so that there is mechanical coupling between these two points. This coupling was included in the NASTRAN analysis through DMIG cards. The hinges were incorporated by releasing the relevant rotational degree of freedom at the outer end of the appropriate member (see Figure 2.1).

An attempt was made to model the drive train precisely although the stiffness and inertia values associated with the gearbox and motor were evaluated. However,

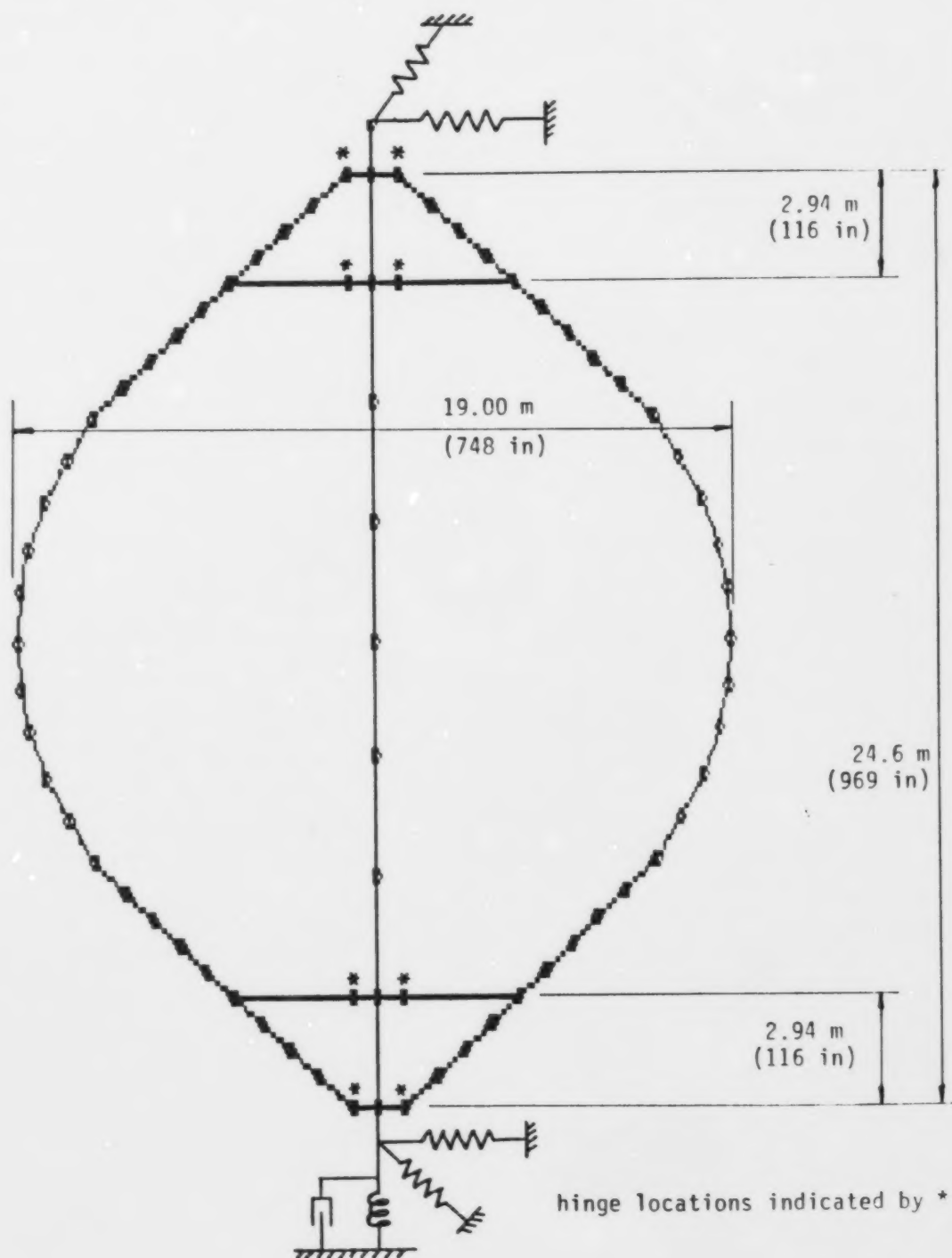


Figure 2.1 Finite Element Idealization of Rotor of Adecon 19-m

the effective viscous damping offered by the motor/generator was included since this may strongly influence torsional modes.

2.3 Natural Frequencies of Adecon 19-m

It has previously been established that rotating frame effects play an important role in determining the natural frequencies and dynamic response of Darrieus rotors. These effects include the stiffening due to centrifugal forces, softening due to whirling, and coupling due to Coriolis forces. The last of these is especially marked in Darrieus rotors due to the flexibility of the blades in both horizontal directions.

The analysis of the Adecon machine was further complicated by the presence of the hinges that allow the blades to rotate about the vertical axes. For the operating rotor, stability is maintained between the aerodynamic and centrifugal forces by the blade advancing through a certain angle. At rated power this angle is approximately equal to 25 degrees for the Adecon 19-m rotor.

Such a large rotation changes the configuration of the rotor considerably. It causes the blade to "toe-in" slightly, thus affecting the aerodynamic forces, and it possibly affects the natural frequencies and modes of vibration. For these reasons it is necessary to identify the mean configuration during operation (at a specified windspeed). This must be done through an iterative procedure since the aerodynamic forces vary with the rotation about the hinge. It was therefore necessary to prepare a FORTRAN program which would carry out this calculation.

Preparation of the program involved the modification of the double multiple streamtube routine to incorporate a toe-in angle. The output from the program includes a set of blade co-ordinates corresponding to the operating configuration and a set of mean and cyclic aerodynamic forces that may be used in a NASTRAN static or dynamic analysis. A listing of the FORTRAN code is included in the separate report on Task #1.

Once the mean configuration was determined, the internal forces due to gravity, centrifugal and mean aerodynamic effects were calculated using a NASTRAN Solution 64 with two subcases. This allowed the differential stiffening matrix to be separately stored. That matrix was then included in a subsequent solution 28 (complex eigenvalue extraction) into which the softening and Coriolis effects were incorporated by DMAP Alters. The differential stiffening matrix had the effect of removing any singularity at the hinges.

The effect of a 25 degree offset (corresponding to a 20 m/s, 45 mph windspeed) is summarized in Table 2.1 which shows that except for the two fundamental frequencies the effect is negligible. These two fundamental frequencies are so low that it was considered that the changes in the natural frequencies would not cause any increase in dynamic amplification of response so that the differences were not significant. All subsequent natural frequency estimates were, therefore, referred to the zero wind (no offset angle) condition.

Figure 2.2 shows how the first eleven natural frequencies vary with rotor speed. The figure includes a

TABLE 2.1 Effect of Windspeed on Natural Frequencies
of Adecon 19-m

Mode No.	Mode Shape	49 rpm	
		V = 0 mph	V = 45 mph
		Hz	Hz
1	P	0.257	0.186
2	B	0.258	0.189
3	2B/FA	1.700	2.683
4	2P/FS	2.746	2.726
5	T/B	3.033	3.029
6	2P/FS	3.677	3.658
7	2B/FA	3.678	3.660
8	B/T	4.549	4.546
9	T2FA/B	5.456	5.458
10	2FS/3P	6.466	6.411
11	2FA/2B	6.473	6.456
12	T2FA/B	7.467	7.458

Notation: P propeller (torsional)
 B butterfly (out-of-plane, in-phase)
 T tower (in-plane motion of column)
 FS flatwise symmetric (in-plane blade motion)
 FA flatwise asymmetric (in-plane blade motion)

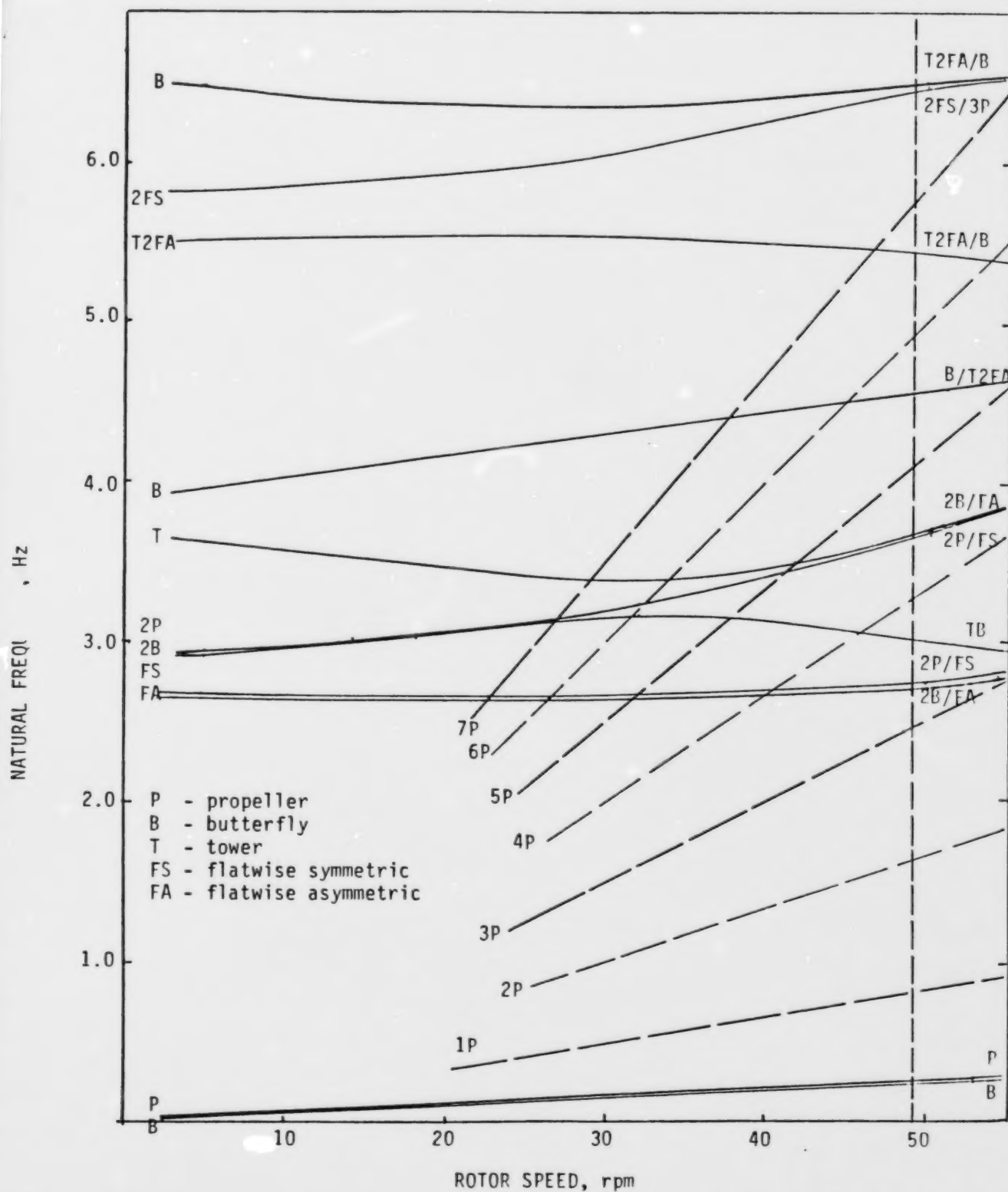


Figure 2.2 Natural Frequencies vs. Rotor Speed
of Adecon 19-m

description of the mode shapes, both at zero rpm and at 49 rpm and the multiples of the rotor speed are also included. The double nature of the modes at 49 rpm is due to Coriolis coupling; the dominant member of each pair is given first.

2.4 Frequency Response of Adecon 19-m

The basic aerodynamic principles of the vertical axis wind turbine imply that in a uniform flow all aerodynamic loads are cyclic with frequencies equal to multiples of the rotational frequency. These multiples or harmonics are referred to as 0P, 1P, 2P, etc. The determination of the aerodynamic loads is dependent on the type and version of the aerodynamic model used and while the performance prediction of different models has been compared, no equivalent comparison of the structural implications is available.

The aerodynamic model used in this study was the double multiple streamtube (DMST) with the facility to interpolate between Reynolds numbers, accommodate different airfoil shapes and chords, and includes a (Masse-Templin) dynamic stall routine. The analysis procedure was similar to that used for the extraction of natural frequencies and is shown schematically in Figure 2.3.

Forced response analyses were carried out for ambient (mid-rotor) windspeeds of 35 and 45 mph (15.6 and 20 m/s). From these two sets of results, it was possible to extrapolate to higher and lower windspeeds. For each windspeed and frequency (from 0P to 5P) the deformed shape was plotted and force resultants (bending moments,

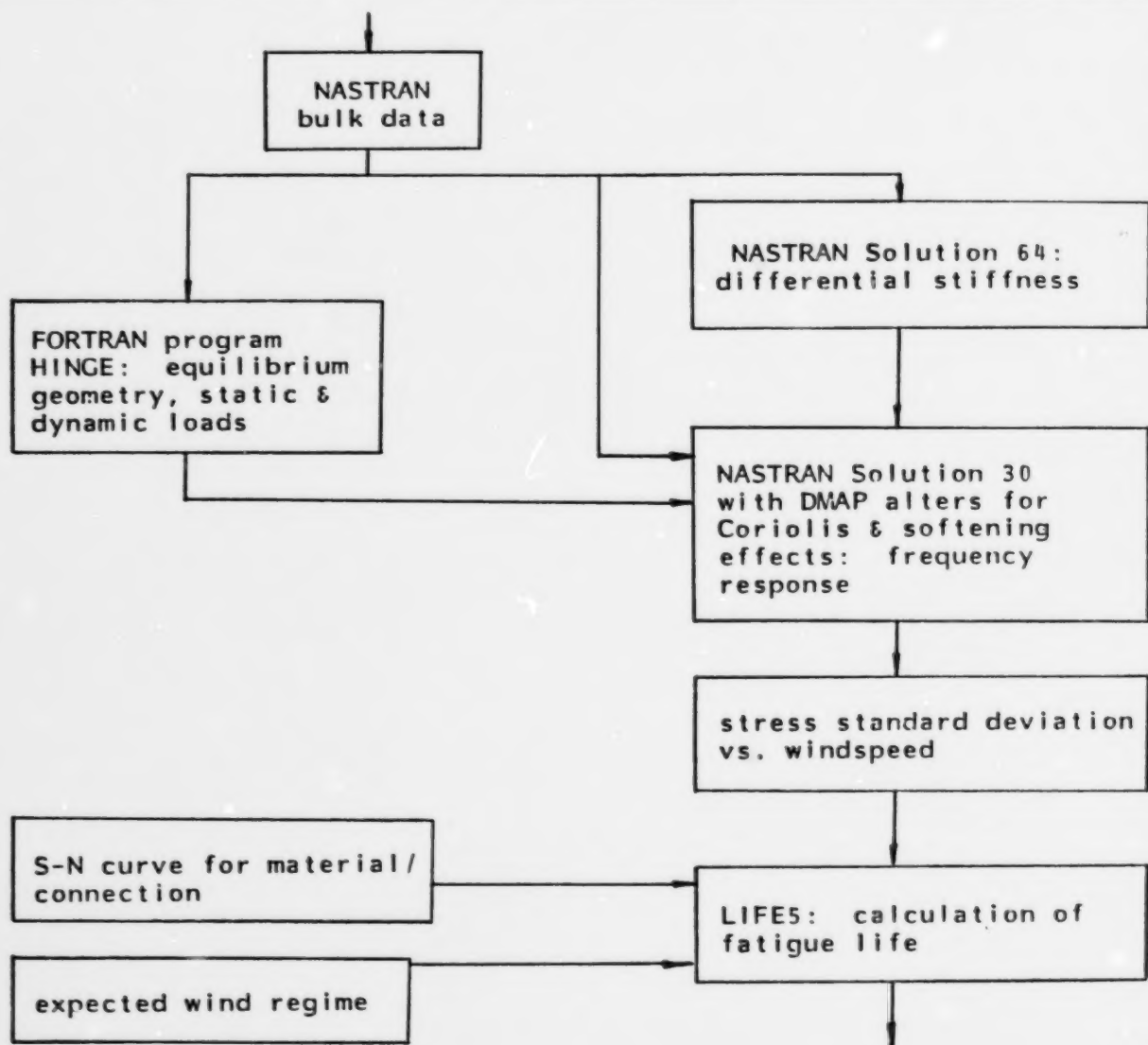


Figure 2.3 Analysis Procedure for Frequency Response

TABLE 2. Summary of Blade and Column Stresses.
Windspeed - 45 mph

	Gravity Only 0P	Centr. Only 0P	Aero. & Centr. 0P	1P	2P	3P	4P	5P	Stan. Dev.
lower root, node 102									
LE	-109	1114	715	678	62	562	87	30	622
TE	-109	1114	1531	680	39	390	121	59	560
IF	-2	821	833	662	18	418	9	2	553
OF	-217	1407	1332	748	22	622	19	23	687
lower strut, node 106									
LE	-41	1208	616	756	174	725	365	155	801
TE	-41	1208	1832	604	205	172	451	200	583
IF	1796	2846	2646	100	140	603	142	158	468
OF	-1879	-428	-319	1304	133	1596	145	168	1468
lower splice, node 108									
LE	-31	1065	624	783	21	635	466	184	796
TE	-31	1065	1508	684	37	694	573	234	816
IF	-855	-682	-576	266	168	592	79	184	491
OF	792	2544	2620	1142	158	401	76	172	871
upper splice, node 122									
LE	31	1065	868	629	148	726	304	96	720
TE	31	1065	1211	680	217	799	367	84	800
IF	861	-682	-531	993	184	81	103	157	728
OF	-798	2544	2576	460	143	909	78	121	734
upper strut, node 124									
LE	41	1208	950	747	308	742	286	60	801
TE	41	1208	1424	767	400	856	335	70	892
IF	-1796	2846	2532	5	333	855	171	143	667
OF	1879	-428	-203	1473	352	157	183	132	1080
upper root, node 128									
LE	109	1114	1022	716	42	493	77	26	617
TE	109	1114	1164	734	90	503	84	13	635
IF	2	821	839	683	16	525	9	15	609
OF	215	1407	1332	763	50	421	28	22	616
central column, node 106									
in-plane	-610	-1100	-1100	2896	0	1988	0	358	2496
out-of-plane	-610	-1100	-1100	2583	0	1538	0	216	2131

Blade Airfoil Locations: LE leading edge
TE trailing edge
IF inner face
OF outer face

etc.) and surface stresses were listed. Table 2.2 presents the mean and cyclic stresses predicted for possible critical locations on the blade in a 45 mph wind.

The combined effects of the predicted cyclic stresses (at 1P to 5P) have been combined as a standard deviation which was calculated as the square root of one-half of the sum of the squares. This relationship is valid for frequencies that are different or have a random phase difference. Such a calculation cannot, with the assumption of uniform flow, include the contribution of any response due to atmospheric turbulence. Recent studies have indicated that turbulence may make significant contributions to in-plane bending.

2.5 Fatigue Life of Adecon 19-m

The fatigue strength of a certain connection depends on many factors such as the material properties, the surface finish, the presence of corrosion, fretting action, and the nature of the load. A common procedure for establishing the expected fatigue life is to first determine the "S-N" curve (cyclic strength vs. number of cycles) and to apply Miner's rule to the range of cyclic stresses predicted. This linear approach to the superposition of fatigue damage is simplistic and can overestimate the fatigue life but is easy to apply and is convenient for preliminary design.

From Table 2.2 it is apparent that the most highly stressed of the blade connections is that with the lower strut. This is not a splice connection but the presence of high stresses in the blade material together with

stress concentrations associated with the fasteners and holes are sufficient to reduce the fatigue strength.

Experience has confirmed that fatigue failure in 6061-T6 aluminum is usually initiated by fretting action of fasteners or adjacent material. The degree of fretting action present directly affects the fatigue strength. Fatigue tests have been carried out by Indal Technologies to determine the strength of root connections on the Indal 6400 which uses 6061-T6 for the extruded blade. These connections may be superior in the quality of details to those on the Adecon machine (which use 6351-T6 alloy) and the fatigue strength may be less.

The combination of the selected S-N curve with the standard deviation of stress at known windspeed and the expected wind regime can be done using a computer program "LIFE" first prepared at Sandia National Laboratories (4). This program has been further developed at Indal Technologies and was used to calculate the fatigue life of the blade connection at the lower strut.

A fatigue strength of 0.75 of the "DAF root tests" was conservatively used and resulted in a fatigue life of 1.91 years if the damage due to start-stop cycles and any response while stationary was neglected. This value was obtained in an 18 mph mean windspeed having a Rayleigh frequency distribution (Weibull shape factor = 2.0). Using an S-N curve corresponding to "DAF root tests * 1.0", the fatigue life increased to 15.2 years and this would be further increased by lower cut-off windspeeds and lower mean windspeed regimes.

3.0 TASK #3: NATURAL FREQUENCIES OF INDAL 6400 WITH HINGED BLADES

3.1 Configuration

There were two essential changes to be made to the mathematical model of the Indal 6400 rotor to incorporate hinges: one was to replace the horizontal struts by extremely flexible members; the other was to modify the upper and lower root connections.

The horizontal struts were replaced by members of nominal flexural stiffness so that they would apply negligible restraint to the blades during operation. Later consideration might require that this stiffness be increased but for this task it was decided to make the blades effectively free.

The location of the blade hinges is important in determining the angle of blade offset during operation. The blades of the Indal 6400 are much heavier than those of the Adecon 19-m rotor and centrifugal tension is much greater. This leads to a smaller offset angle being required for equilibrium during operation. The FORTRAN program HINGE was used to calculate this angle and the relationship between hinge eccentricity and maximum blade offset angle is summarized in Figure 3.1. This figure confirms the approximately linear relationship between these two variables at any selected windspeed.

A hinge eccentricity of 55 in. (1.40 m) was selected for Tasks #3 and 4 since this corresponds to a maximum blade offset of 9.5 degrees. The finite element model of the Indal 6400 rotor was then modified to incorporate

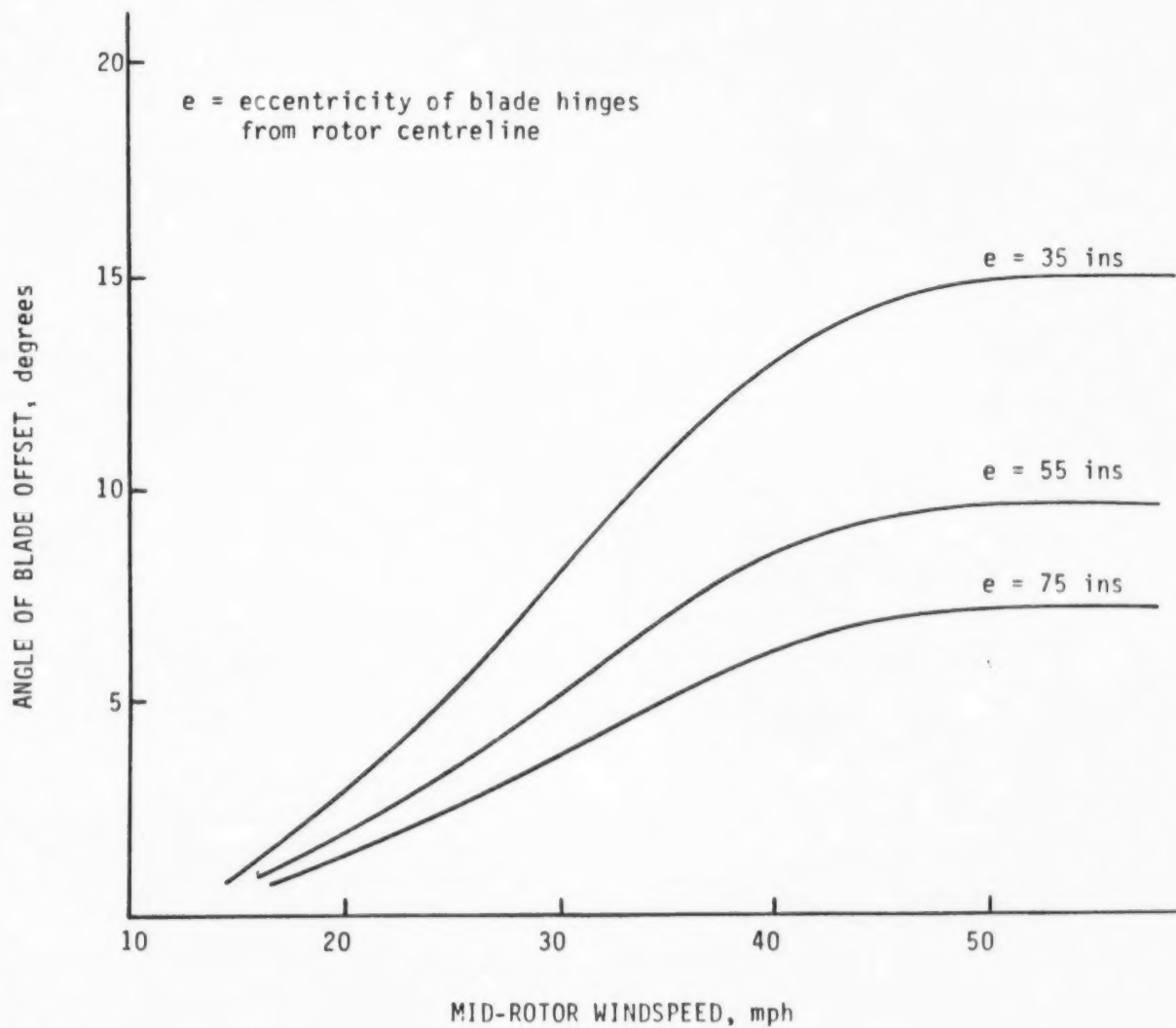


Figure 3.1 Effect of Blade Eccentricity and Windspeed
on Blade Offset Angle of Modified Indal 6400

both vertical and horizontally oriented hinges in a manner similar to the Adecon 19-m. The outline of the Indal rotor is shown in Figure 3.2.

3.2 Analysis Procedure

The extraction of natural frequencies of the Indal 6400 with blade hinges followed the same steps as for the Adecon 19-m rotor:

1. The differential stiffness matrix was generated and stored using a NASTRAN solution 64 (non-linear statics).
2. A NASTRAN solution 28 (complex eigenvalue extraction) was carried out with the addition of the differential stiffness matrix together with the Coriolis and softening matrices. This solution was carried out at a number of rotor speeds.

Earlier work on the Adecon 19-m rotor established that the blade offset angle (of up to 25 degrees) made little difference to the natural frequencies. Therefore the underformed configuration of the Indal 6400 was used throughout Task #3.

3.3 Numerical Results

The natural frequencies of the Indal 6400 with hinged blades are summarized in Figure 3.3 which shows how the first twelve natural frequencies vary with rotor speed. This "fan plot" can be compared with the equivalent plot for the Adecon 19-m (Figure 2.1) and with the

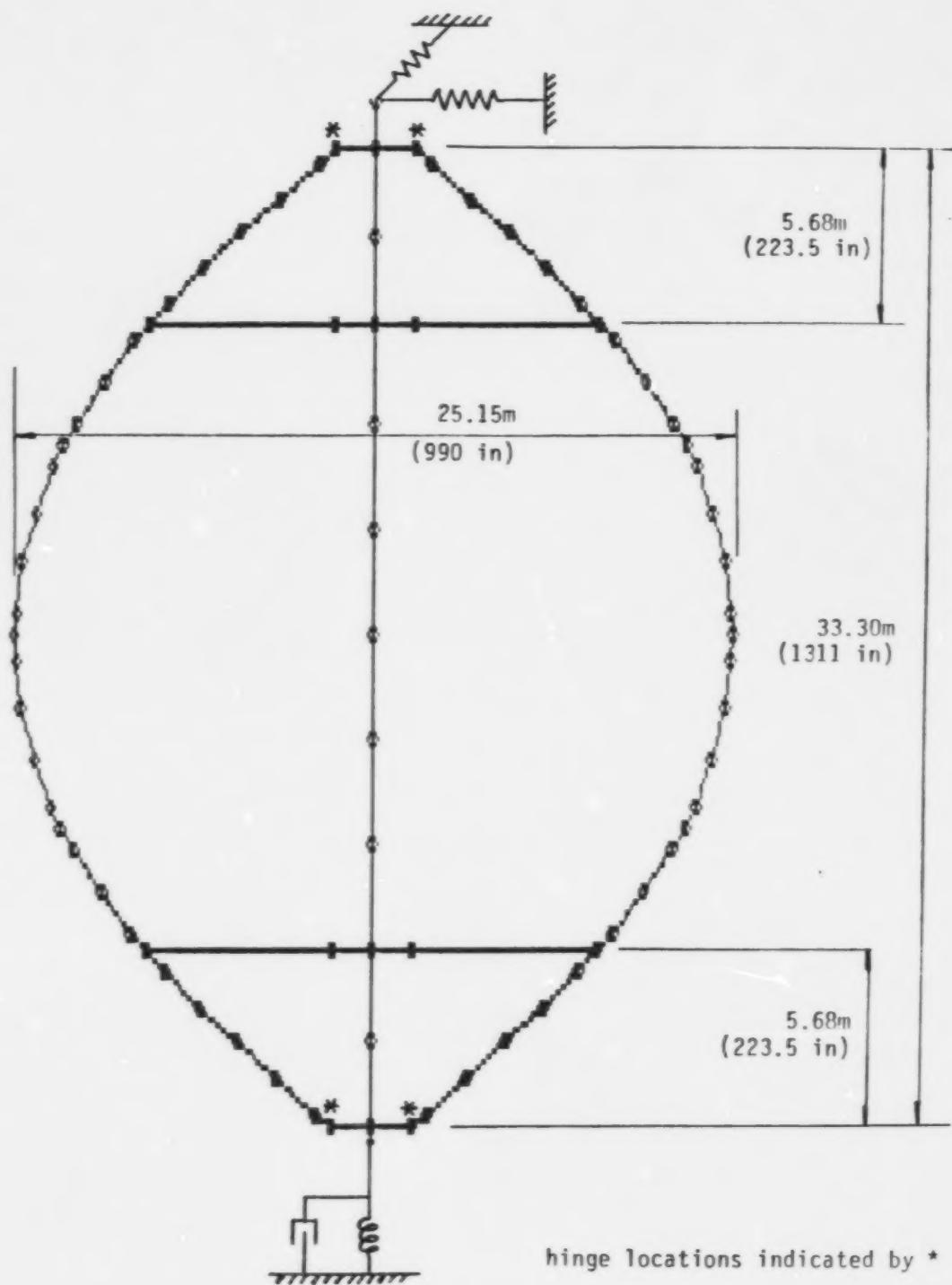


Figure 3.2. Outline of Indal 6400 with Hinges

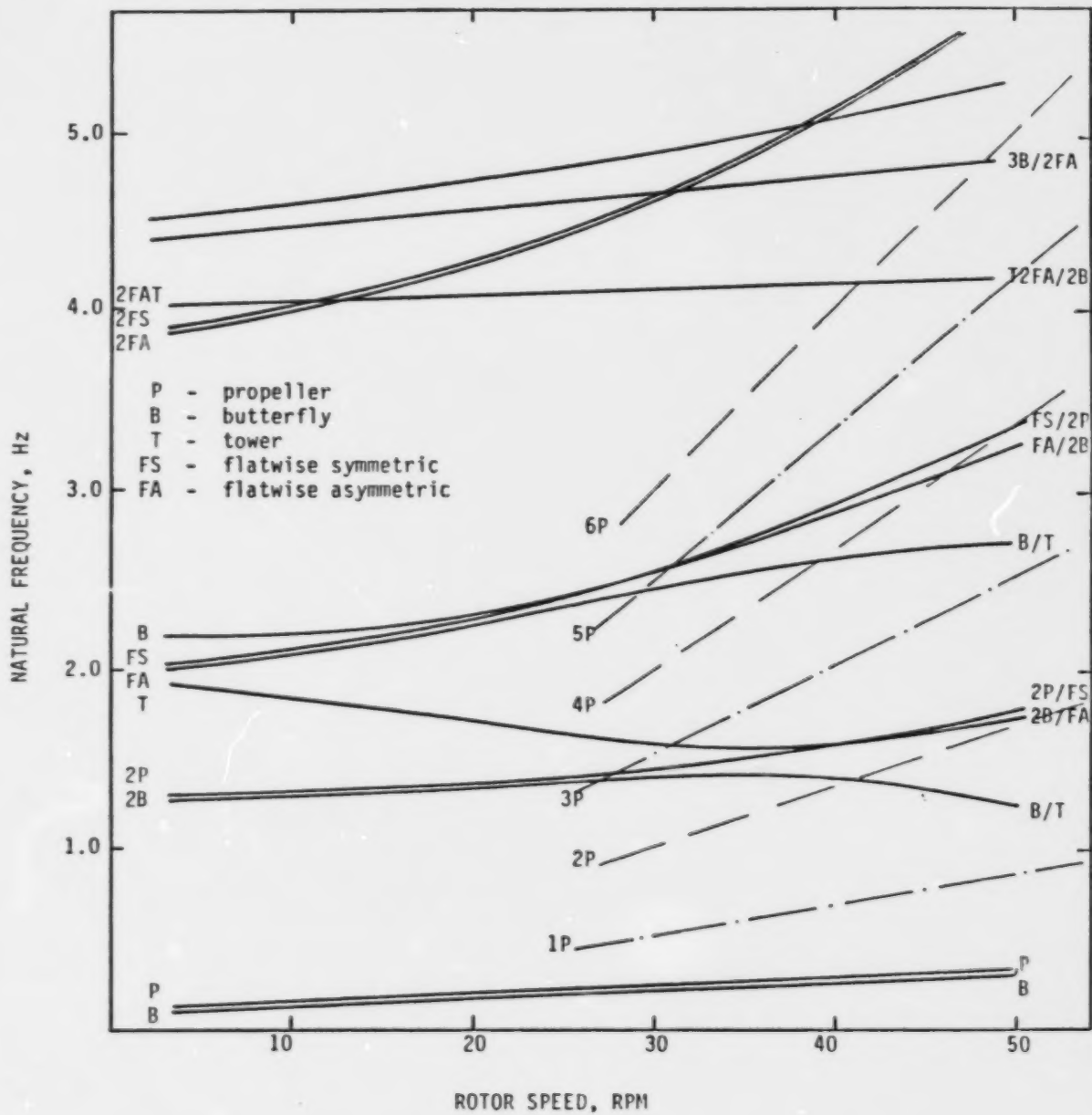


Figure 3.3 Indal 6400 with Hinged Blades.
Natural Frequencies vs. Rotor Speed

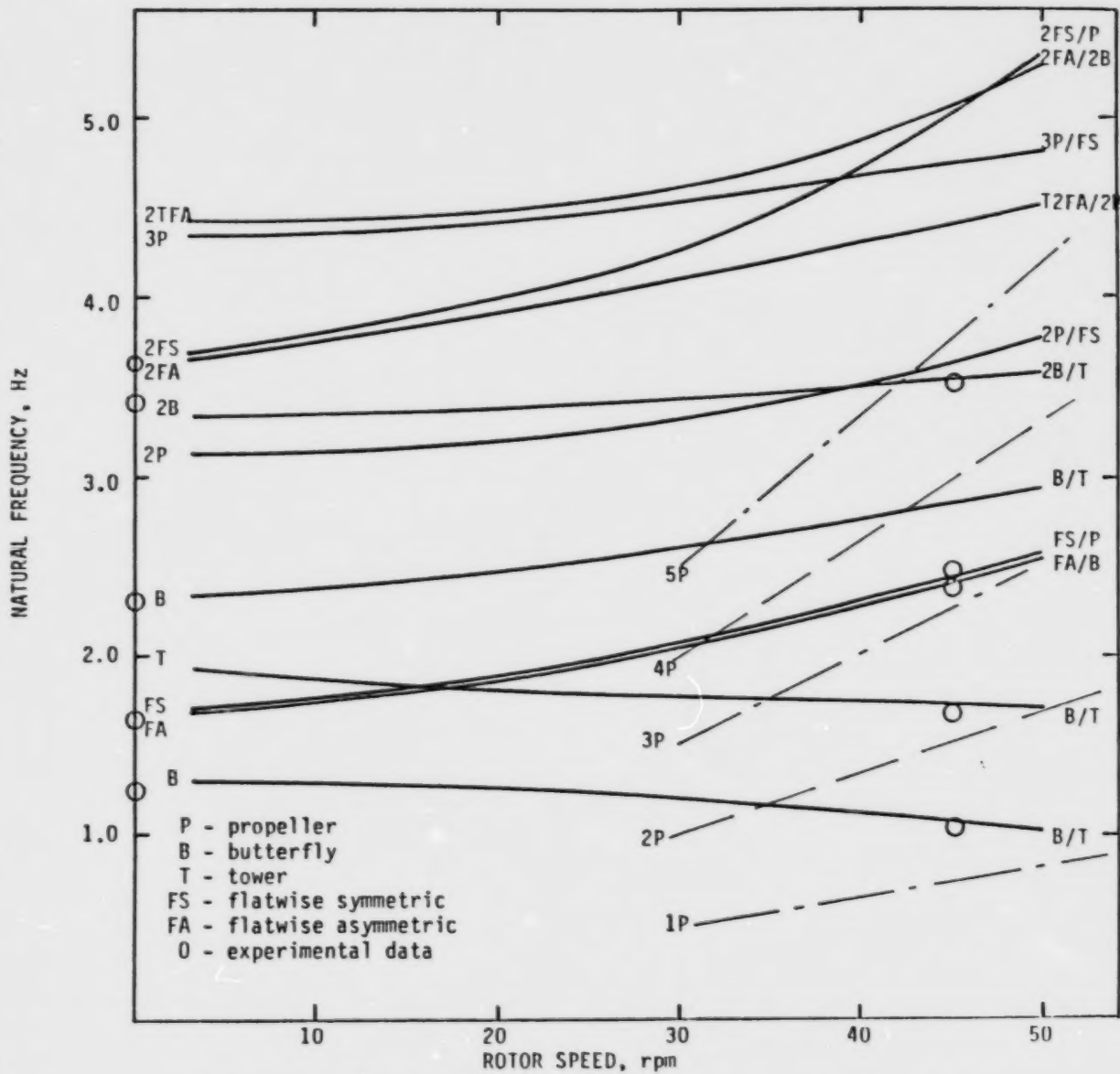


FIGURE 3.4. Indal 6400 (PEI mk.2) without Hinges: Natural Frequencies vs. Rotor Speed

plot for the Indal 6400 before the addition of hinges (Figure 3.4), such as for the present machine on PEI. The mode shapes of the Indal 6400 with hinges at 45 rpm are included as Appendix A.

In the same manner as the Adecon 19-m, the Indal 6400 with hinges has a pair of low-frequency fundamental modes. Both fan plots also show that the second butterfly mode has a lower frequency than the first butterfly mode (at 0 rpm). Both fan plots are also characterized by the pseudo-crossing of one or more modes by another; in fact, one complex mode changes place with another.

4.0 TASK #4: FREQUENCY RESPONSE OF INDAL 6400 WITH HINGED BLADES

4.1 Analysis Procedure

The procedure followed for the frequency response of the Indal 6400 with hinged blades was identical to that used for the Adecon 19-m rotor (see Figure 2.3).

The analysis is applicable to steady-state operation of the rotor in a uniform flow field and the response therefore appears at multiples of the rotor speed only. The effects of atmospheric turbulence have, therefore, been neglected; this approximation may cause an underestimate of the flatwise (in-plane) response of the blades.

4.2 Numerical Results

The frequency response analysis was performed for ambient windspeeds (at mid-rotor) at 30 and 45 mph (13.4 and 20.1 m/s). Some relevant results from these analyses are presented in Tables 4.1 and 4.2. These tables list the mean and cyclic stresses at the critical locations on the blade - found to be at the upper and lower strut connections. For comparison the stresses adjacent to the lower root are also listed.

TABLE 4.1 Blade Stresses in Indal 6400 with Hinges.
Windspeed = 30 mph (13.4 m/s)

Location	0P		Cyclic Response					rms of Cyclic
	Gravity Only	Centrif. Only	Aero Only	1P	2P	3P	4P	5P
lower root (node 101)								
leading edge	-157	2316	-92	73	37	12	8	5
trailing edge	-157	2316	70	44	89	11	7	9
inner face	-548	2829	30	54	102	12	13	8
outer face	232	1796	-61	67	14	11	9	2
lower strut (node 106)								
leading edge	-45	1978	-259	143	300	50	30	36
trailing edge	-45	1978	267	66	366	67	38	45
inner face	2319	2033	-98	573	72	29	105	23
outer face	-2409	1923	67	689	71	33	115	21
upper strut (node 124)								
leading edge	44	1978	-180	243	121	15	17	30
trailing edge	44	1978	185	254	150	14	18	40
inner face	-1433	2073	-101	522	136	92	154	31
outer face	1523	1882	76	617	132	93	165	35

* Stress values are given in psi (MPa * 0.00689)

* Cyclic responses are amplitudes.

TABLE 4.2 Blade Stresses in Indal 6400 with Hinges.
Windspeed = 45 mph (20.1 m/s)

Location	0P		Cyclic Response								rms of Cyclic
	Gravity Only	Centrif. Only	Aero Only	1P	2P	3P	4P	5P			
lower root (node 101)											
leading edge	-157	2316	-115	100	118	31	44	35			118
trailing edge	-157	2316	86	72	129	26	7	35			109
inner face	-548	2829	47	110	164	31	46	16			145
outer face	232	1796	-90	58	29	21	14	10			68
lower strut (node 106)											
leading edge	-45	1978	-328	186	394	104	90	198			352
trailing edge	-45	1978	342	159	458	115	73	228			409
inner face	2319	2033	-136	709	363	457	580	171			777
outer face	-2409	1923	88	848	383	508	615	182			876
upper strut (node 124)											
leading edge	44	1978	-255	163	709	39	166	183			544
trailing edge	44	1978	258	214	890	17	208	233			684
inner face	-1433	2073	-148	994	623	332	280	161			892
outer face	1523	1882	101	1152	657	347	311	176			1002

* Stress values are given in psi (MPa * 0.00689)

* Cyclic responses are amplitudes.

5.0 TASK #5: TRANSIENT RESPONSE OF INDAL 6400 WITH HINGED BLADES

During normal operation an equilibrium exists between the centrifugal forces on the blade and the mean aerodynamic forces. This equilibrium does not exist during the transient periods of starting and stopping and without additional restraint the blade would undergo large rotations and cause serious damage. To control the deformation due to these transient forces the Adecon 19-m rotor is equipped with a system of hydraulic dampers. The present study proposed to examine how the blades of the Indal 6400 might be stabilized by the use of flexible struts.

5.1 Survival Wind Loading

The existing Indal 6400 has been shown (mathematically) to remain stable and not be overstressed under survival wind loading. This analysis considered the large deformation of a blade which was orientated directly upwind and was restrained at the root and strut connections. The analysis did not consider possible aeroelastic instabilities (such as flutter) of the parked rotor.

The in-plane behaviour of the modified Indal 6400 (with hinged blades) under in-plane loading only will be very similar to that of the existing 6400. Any effective horizontally orientated hinges at the roots will reduce the flexural restraint in those regions but will not significantly reduce the "buckling" load of the entire blade. No additional examination of this case was, therefore, considered necessary.

Any out-of-plane loading of the blades will cause considerable blade displacements unless some additional restraint is offered to the blades. This may be supplied by the additional, stiffer struts which are discussed further in Section 5.3. Even if the blades are restrained from out-of-plane motion, there is some chance of aerodynamically-induced vibrations occurring. However, an investigation of this phenomenon is beyond the scope of the present report.

5.2 Critical Conditions

The most critical transient condition is likely to correspond to full braking on a rotor which is rotating slowly. In this condition there is little or no stabilizing influence from centrifugal forces. The magnitude of the (relative) motion and forces induced by braking will depend on the value of the braking torque applied and the manner in which it is applied. For this study typical (hard) braking of the Indal 6400 was used. This was approximated by a linear increase over 0.5 seconds to 2.0 of the rotor torque at rated power. Thereafter the torque remained constant at 187,500 ft.lb (256,000 N.m).

5.3 Configuration

The problem faced in this task was how to design a set of horizontal struts that would be sufficiently flexible to allow effectively free blade motion during normal operation but stiff and strong enough to restrain the blades during transient loads. In order to make this problem more tractable the hinge eccentricity was increased from 55 to 75 inches (1.40 to 1.90 m). This

reduced the maximum blade offset during operation from 9.5 to 7.0 degrees (see Figure 3.1) and thereby reduced the flexibility requirements on the struts.

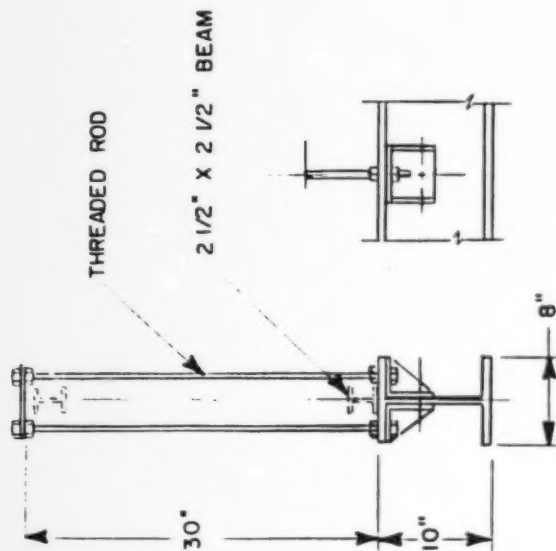
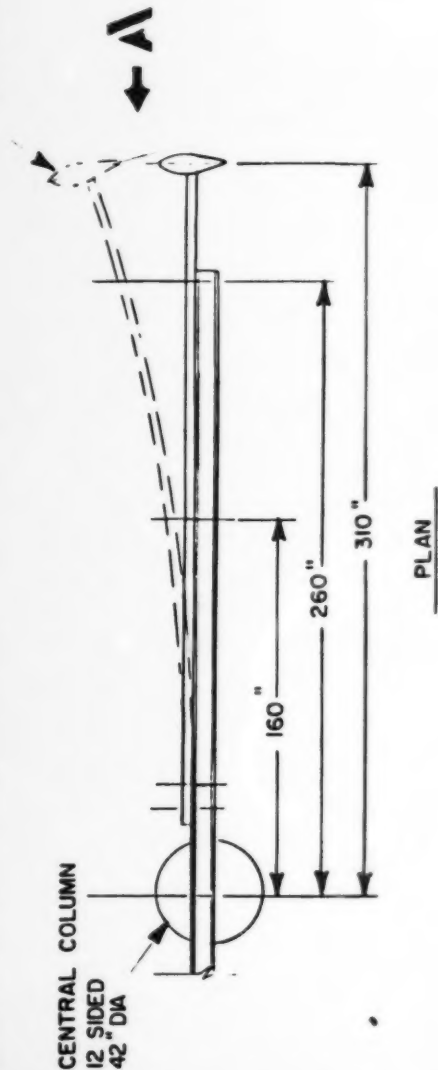
In order to carry out a preliminary assessment of the problem it was assumed that during normal operation the out-of-plane reaction between strut and blade should not exceed 100 lbf (448 N). This implies that the struts transmit not more than 11% of the total rotor torque (approximately). The tip displacement of the strut under these conditions is approximately 28 inches (0.711 m) and implies a (uniform) flexural stiffness of $34.E6 \text{ lb.in}^2$. If a steel member were used then a 2.5 x 2.5 inch I-beam would be adequate and would result in maximum bending stresses of 30,000 psi (206 MPa).

Alternative materials such as glass fibre or more exotic composites could be used in place of steel although their traditional advantage of high strength-to-weight ratio is not important in this application. A typical fibre glass-epoxy composite may have an ultimate strength of up to 100,000 psi (689 MPa) and an elastic modulus of $6.E6 \text{ psi}$ (41,000 MPa). Such a material could certainly be used as a flexible strut for normal operation. However the main hurdle for the designer is the behaviour under braking loads. In Section 5.2 the maximum braking torque was limited to 2.0 of the rated power torque and preliminary design assumed a dynamic amplification of 2.0. The effective static torque was, therefore, 4.0 of the rated power torque.

This braking torque corresponds to a very high lateral load on the strut since little or no torque can be transmitted through the blade roots. The resistance

REVISIONS			DATE	APPROVED
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**CONFIGURATION
AT RATED POWER**



VIEW ON ARROW

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QTY REQD		FSCM		PART OR IDENTIFYING NO	
QTY REQD		FSCM		NOMENCLATURE OR DESCRIPTION	
REQUIREMENTS - UNLESS OTHERWISE SPECIFIED		LIMITS		CONTRACT NO	
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Figure 5.1 Preliminary Design of Strut Arrangement

of the struts to lateral loads must be highly non-linear with a sharp increase in stiffness once the displacement exceeds the normal operating range. One simple arrangement for achieving this is sketched in Figure 5.1 which shows a more solid steel I-beam backing up the flexible strut.

5.4 Analysis Procedure

The dynamic behaviour of this two-step strut is complex and depends upon any energy absorption during impact between the two struts. However, an analysis was attempted in order to give an indication of its feasibility and of further problems. It was assumed that the rotor was moving slowly in zero wind conditions (zero blade offset and zero rotating frame effects). A transient analysis (NASTRAN Solution 27) was carried out on the rotor with the flexible strut free of the stiffer strut. This analysis defined the initial relative motion between the two sets of struts and defined the motion at the time of impact (when the separation between the two reached 27 inches, 0.686 m, at the end of the stiff strut). This motion is described by a series of displaced shapes which are included as Appendix C.

The configuration and velocities at the time of contact between the flexible and stiff struts was noted and used as initial geometry and velocity in a second transient analysis (NASTRAN Solution 27). In this second transient analysis the two sets of struts were connected by rigid rods at the tip of the stiff struts and at a position half way along the stiff strut. The displacement (at the lower strut level) that subsequently occurred is traced in a series of plots attached as Appendix D.

5.5 Numerical Results

Table 5.1 lists the bending stresses in some critical locations of the flexible and stiff l-beams at various times after initiation of braking. These results correspond to the firm attachment between the two struts at radii 160 and 260 inches once contact at radius 260 inches has occurred. This is a crude simulation but can be of use in pointing to potential problems.

Until impact occurs the stresses in the two struts increase as expected - the root stresses in the flexible strut increase rapidly as the relative displacement at its tip increases. After impact the flexible strut is bent severely at its outermost point of restraint (at radius 260 inches) while the bending stresses elsewhere do not rise so sharply. In the stiff strut the root bending stresses show a maximum soon after impact.

In the blade the lead-lag stresses are modest prior to impact but peak at approximately 16,000 psi (110 MPa) soon after impact.

TABLE 5.1 Transient Stresses In Lower Strut System
Due to Braking

time after start of braking seconds	root	flexible strut radius = 160 node 301	radius = 260 in. node 302	stiff strut, root	blade at lower strut trailing edge
0.0	0	0	0	0	0
0.05	657	144	123	1156	88
0.10	1721	207	198	1686	144
0.15	4524	819	355	2864	209
0.20	9035	2345	335	3561	208
0.25	16307	4798	383	4659	215
0.30	26541	8645	296	4840	167
0.35	40899	13656	320	5959	101
-----strut - strut impact-----					
0.375	55507	-849	88381	13317	-4441
0.400	54655	-50	82258	42663	-9438
0.425	52504	03520	88182	23865	-14612
0.450	55062	4639	46816	15181	-15900
0.475	46895	18922	-15876	32973	-7368
0.500	47111	7241	22904	15213	110

Units: stresses are in psi (MPa * 0.00689)

6.0 SUMMARY & CONCLUSIONS

The summary of the various tasks is divided into three categories: natural frequencies, frequency response, and transient response. Unless noted otherwise, they refer to both the Adecon 19-m and to the modified Indal 6400.

6.1 Natural Frequencies

1. The influence of the windspeed and the offset of the blade at the hinges has little influence on the natural frequencies except for the two fundamental modes. The change to these modes will not appreciably change any forced response during operation.
2. Except for the low value of the two fundamental natural frequencies, the fan plots are similar to those for other Darrieus rotors. This is because the in-plane modes are not affected by the hinge action. The presence of the hinges in no way reduces the possibility of dynamic response in the form of these modes.
3. The action of the hinges and the resulting out-of-plane compliance of the blades leads to Coriolis coupling that is even more apparent than in conventional Darrieus rotors.
4. The two fundamental natural modes involve essentially rigid body rotation of the blades about the hinges.

5. The presence of the hinges at the root and strut level renders the blade assembly to be torsionally soft and causes the second propeller and second butterfly modes therefore to be of low frequency.
6. Comparison of modes 3 and 7 and modes 4 and 6 of the Adecon 19-m at 49 rpm show these pairs to be very similar in shape although having very different frequencies. Examination of the complex modes reveals that in each pair one component is exactly (180°) out-of-phase with the other. For example, while in one the out-of-plane motion may follow the in-plane motion by 90° , in the other mode the same motion will precede the other by 90° .

6.2 Frequency Response

1. The torque ripple is insignificant.
2. The presence of the hinges also attenuates the (butterfly) response at 1P for out-of-plane (edgewise) blade bending.
3. The in-plane (flatwise) blade bending stresses at 1P are not significantly attenuated and may even increase.
4. The presence of the hinges causes attenuation only in those out-of-plane modes that involve rigid body rotation about the hinges. There are, therefore, considerable stresses at frequencies 3P, 4P, and 5P, especially in the in-plane or flatwise modes.

5. The greatest cyclic stresses in the Adecon 19-m occur at 3P due to flatwise bending adjacent to the lower strut and the maximum rms of cyclic stress at 45 mph is 1468 psi (10.1 MPa) on the outer face adjacent to the lower strut.
6. The 1P and 3P axial stresses contribute appreciable to the total rms stress. This is due to the lightweight construction of the Adecon blades.
7. The greatest cyclic stress in the modified Indal 6400 occurs at 1P in flatwise bending. The highest rms of cyclic stresses in a 45 mph windspeed is 1002 psi (6.91 MPa) on the outer face adjacent to the lower strut.
8. The fatigue life of the Adecon 19-m was conservatively estimated to be between 2 and 15 years due to operating conditions in a 18 mph mean Rayleigh wind regime. The exact value depends on the fatigue strength of the blade connections.
9. The fatigue life of the modified Indal 6400 (with hinges) in the same environment is estimated to be over 100 years.
10. The corresponding maximum rms stress in the unmodified Indal 6400 (as installed recently in PEI) is predicted to be 1300 psi (8.97 MPa). The reduction in peak rms stress due to the introduction of hinges is, therefore, approximately 30%.

6.3 Transient Response

1. While response during operation may be decreased by the inclusion of blade hinges, the response to braking threatens to be a serious design condition. Full braking on a rotor which is moving slowly may be the critical condition.
2. In order for the struts of the modified Indal 6400 to satisfy both the flexibility requirements of normal operation and the strength requirements of the braking load, a very nonlinear response is necessary.
3. While high strength-to-weight composites may offer the flexibility required they do not hold any strength advantages over the higher strength structural steels.
4. A possible configuration was examined in which a flexible (steel) strut becomes supported by a stiffer (steel) strut once displacements exceed the normal operating range. This analysis, carried out in two stages, did not consider possible loss of energy during impact or the precise mechanics of rebound. However, it did indicate high transient loads in both strut components and in the blade.
5. The maximum stress in the flexible strut was predicted to be 88,000 psi, in the stiff strut 42,000 psi, and 16,000 psi in the blade. These stresses are equivalent to a very high dynamic amplification and, while they might be carried by higher strength steels, they are not acceptable in the aluminum

blade. They point to the importance of careful softening of the shock loads associated with braking together with gradual application of the brake torque.

6. A difficulty associated with the nonlinear strut requirement is that the central column, having only a modest rotational inertia, decelerates rapidly and exacerbates the final deceleration of the blade once the stiffer strut comes into play.

6.4 Conclusions

The overall conclusion of this study is that while incorporation of vertically orientated hinges in the roots of a Darrieus rotor blade may reduce the rms of cyclic stresses by over 30%, there are significant problems.

One problem, which has not been discussed, is the engineering and reliability of suitable hinges. Another problem, which has been partially explored in this study, is the structural response during transient loading, especially during braking. Flexible struts must be offered additional restraint during braking and any sudden changes in structural response may result in high stresses.

The inclusion of blade hinges does result in a negligible torque ripple in the drive train and power output. However, there may be alternative methods of reducing this feature such as high-slip motors or the increasing of the total rotor inertia. Moreover, the

extent of torque ripple does not normally govern the design of either rotor or drive train and in a windfarm of many machines the total power supplied is not affected by torque ripple.

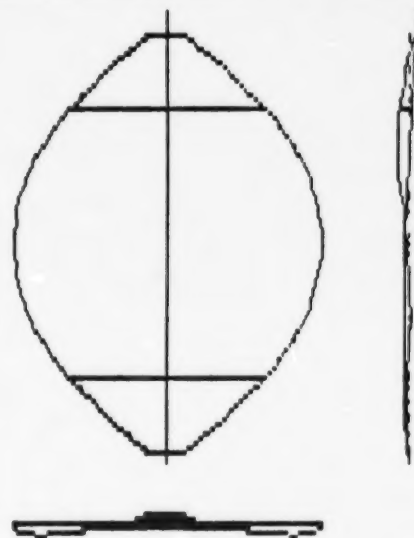
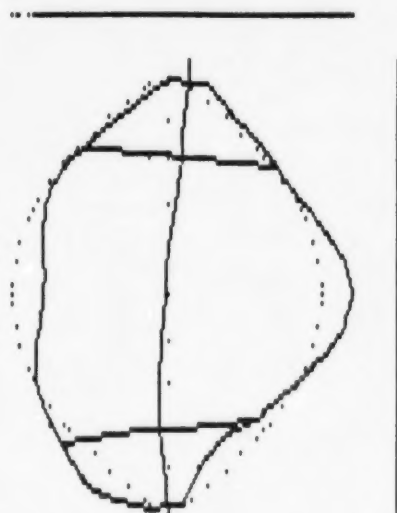
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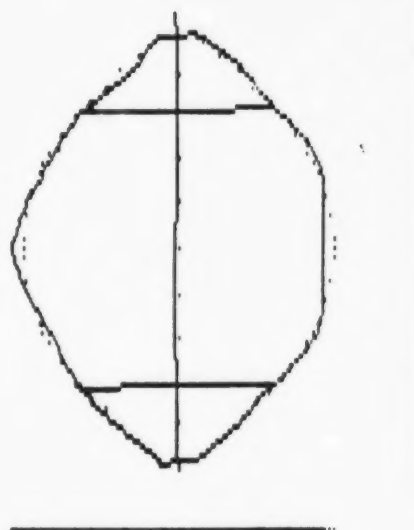
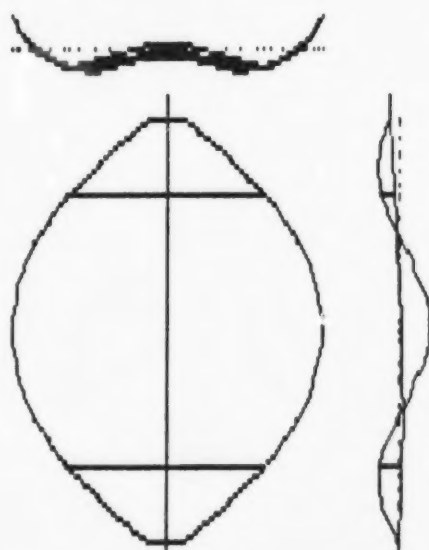
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4. P.S. Veers, "Blade Fatigue Life Assessment with Application to VAWTs", ASME Journal of Solar Energy Engineering, Vol. 104, May 1982, pp. 106-111.
5. D.J. Malcolm, "Fatigue Life Assessment of VAWT Components", Canadian Wind Energy Conference, Ottawa, Dec. 9-10, 1985.
6. "Development of a Hinged Blade for Quality of Power and Fatigue Life Improvement of the Adecon 125 VAWT", SSC Contract OSZ-85-00052, October 1985.
7. "Structural Modelling of Darrieus Rotor Wind Turbines with Hinged Blades and Stabilizing Struts", Indal Technologies Inc. on behalf of Supply and Services Canada, Contract 23216-6-6244, Sept. 1987.

APPENDIX A

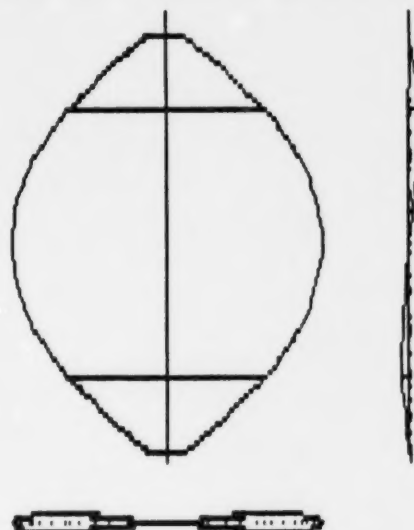
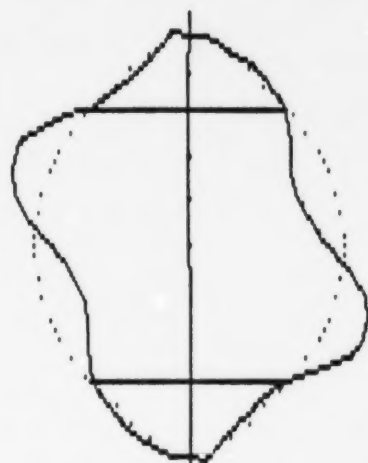
Natural Modes of Vibration at 45 rpm of
Indal 6400 with Hinges



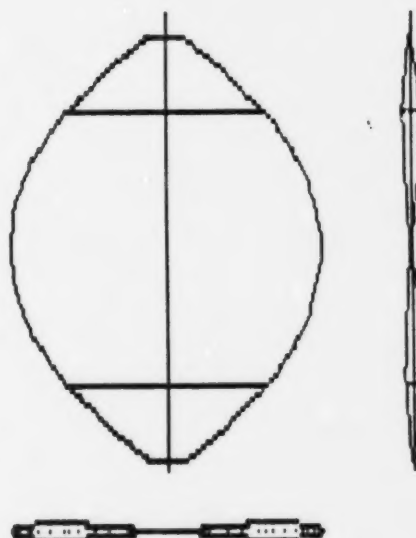
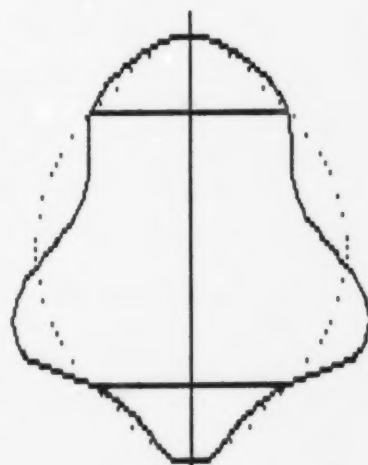
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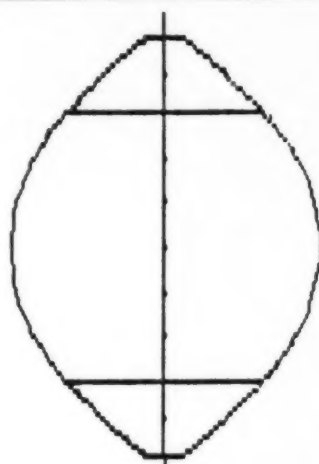
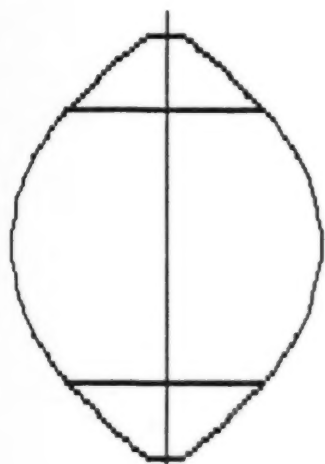
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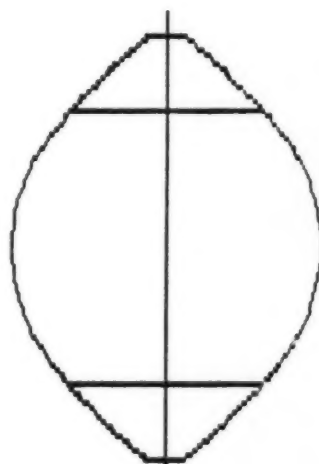
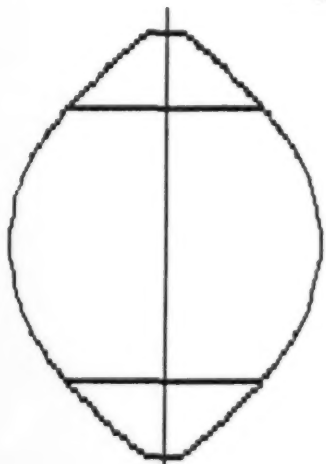
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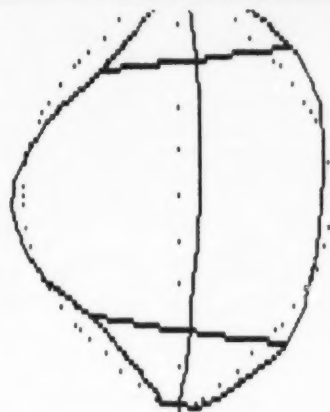
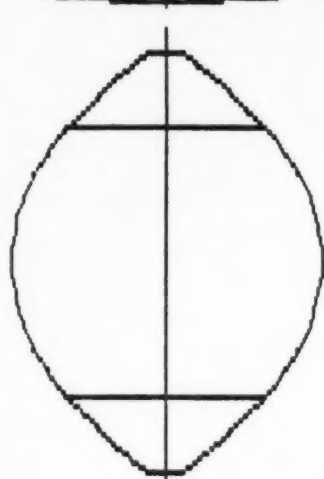
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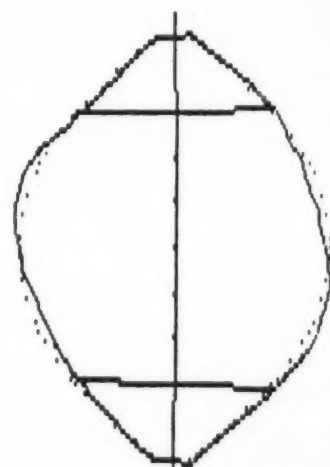
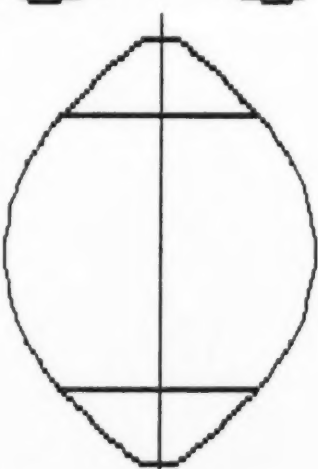
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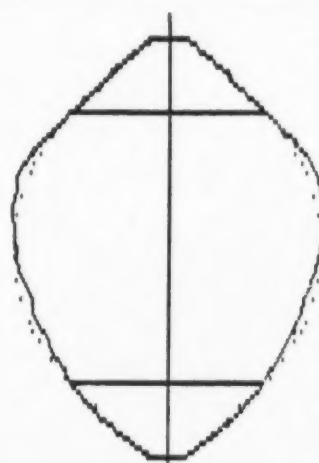
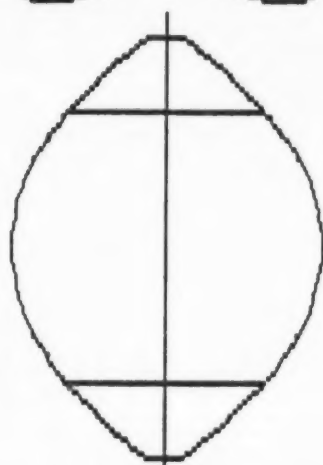
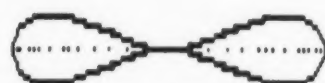
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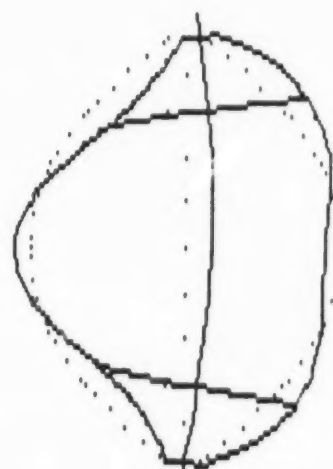
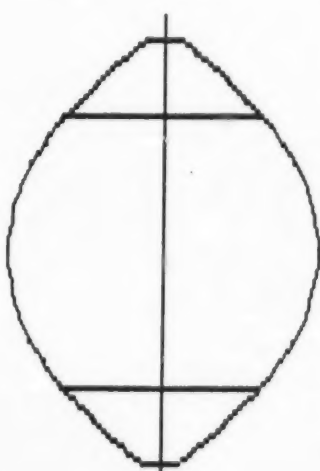
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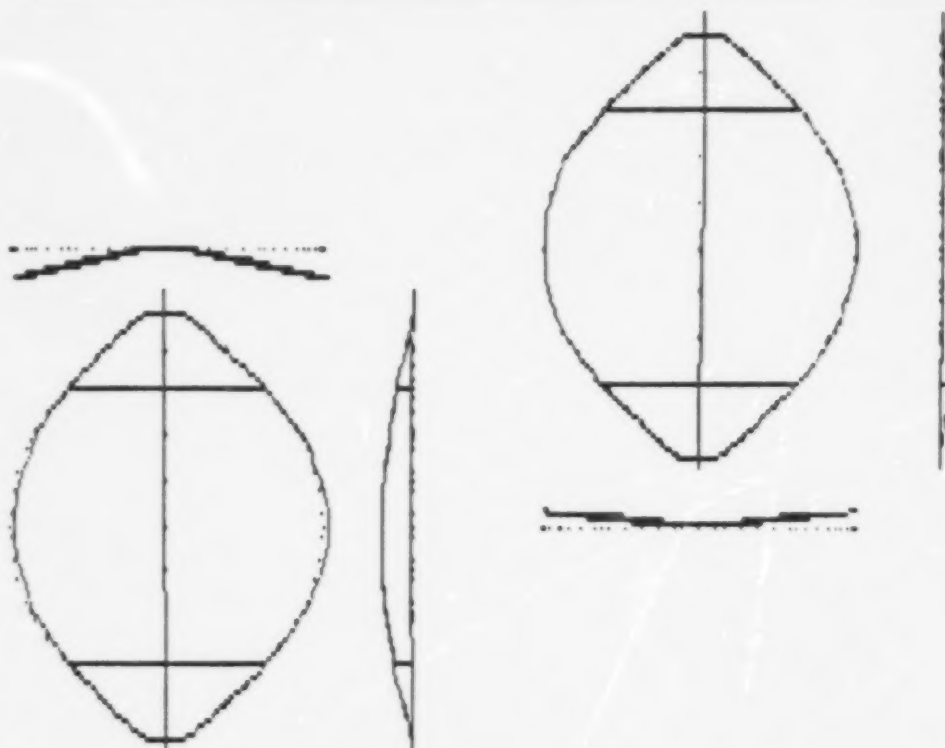
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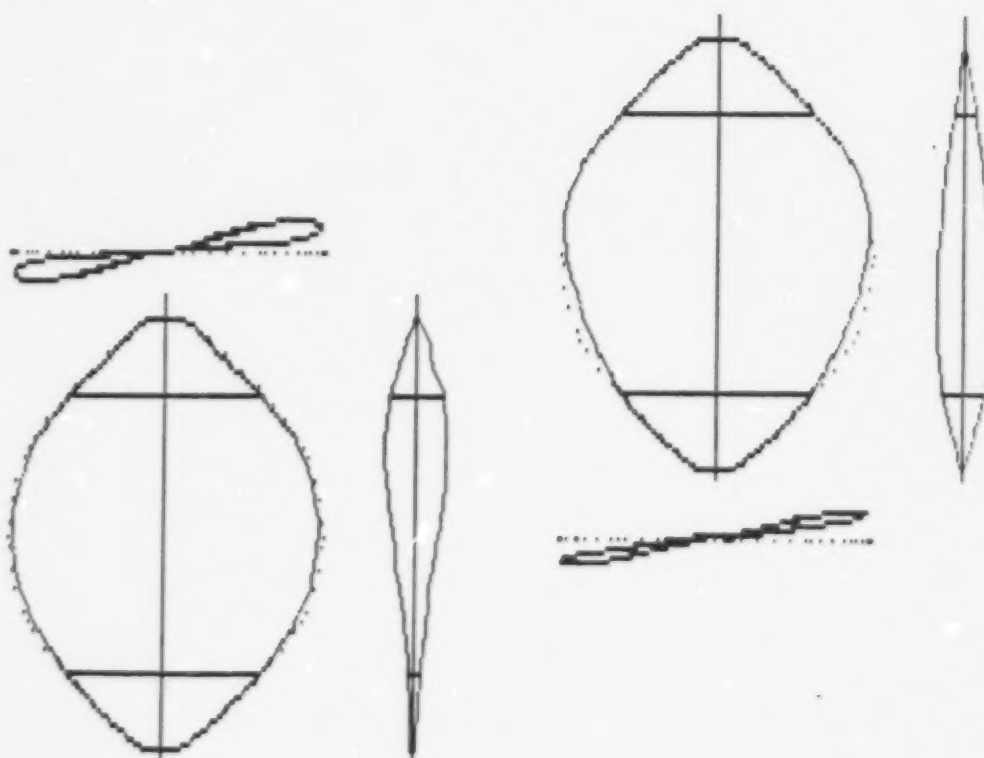
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APPENDIX B

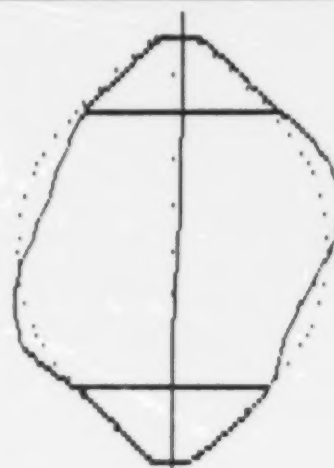
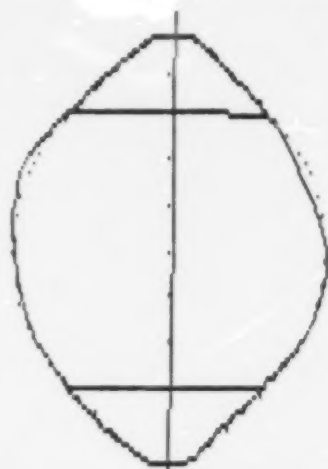
Frequency Response of Indal 6400 with Hinges.
Displaced Shapes in a 45 mph (20.1 m/s) Windspeed



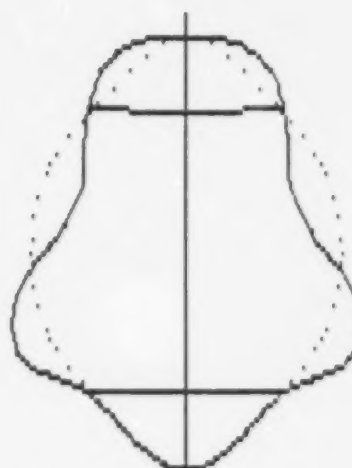
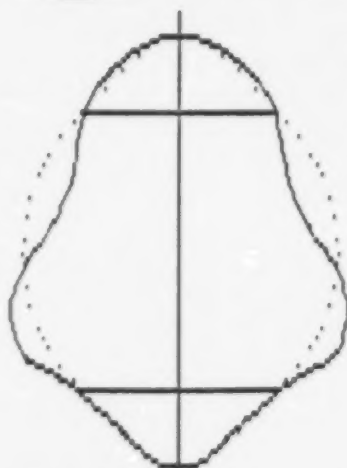
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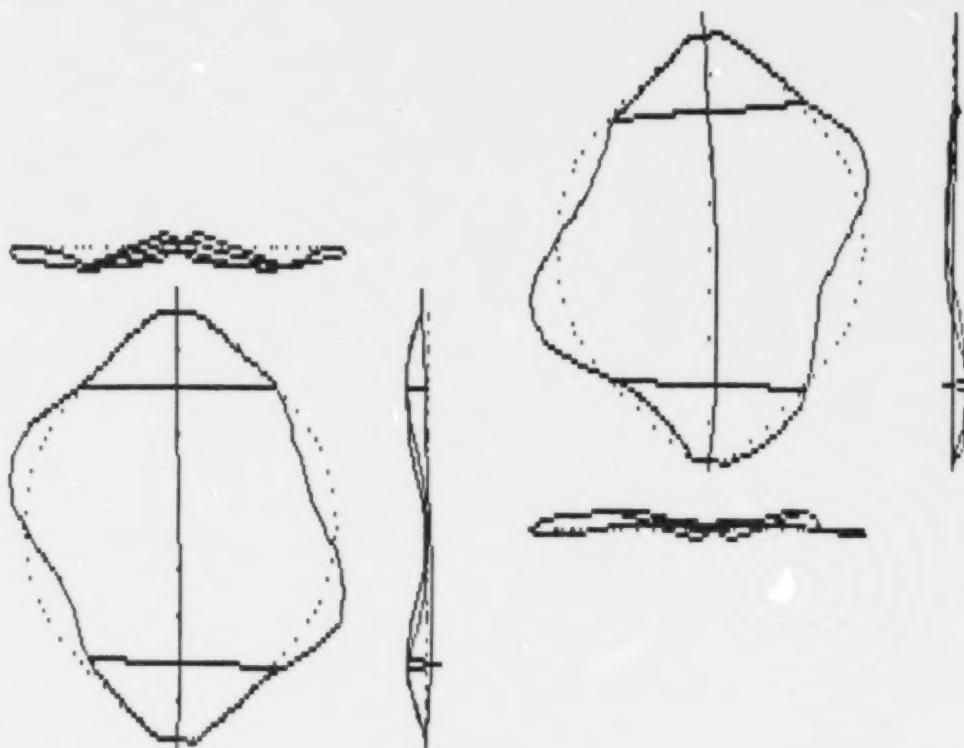
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HINGED 6400 45RPM 45MPH 11AUG87 FREQUENCY= 2.25 HZ MAX DISPL= 0.59 INS



HINGED 6400 45RPM 45MPH 11AUG87 FREQUENCY= 3 HZ MAX DISPL= 0.30 INS



HINGED 6400 45RPM 45MPH 11AUG87FREQUENCY= 3.75 HZ MAX DISPL= 0.14 INS

APPENDIX C

Transient Behaviour of Indal 6400 with Hinges under Braking Loads. Initial Displacement at Lower Strut Level

NOTE (1) These displaced shapes, which are multiplied by a factor of 10.0, are plan views of the lower strut assembly only.

(2) The dotted lines indicate the initial configuration.

(3) The braking torque was increased linearly from zero to two times rated power torque (187,500 ft.lbs., 256,000 N.m) in 0.5 seconds and thereafter remained constant.

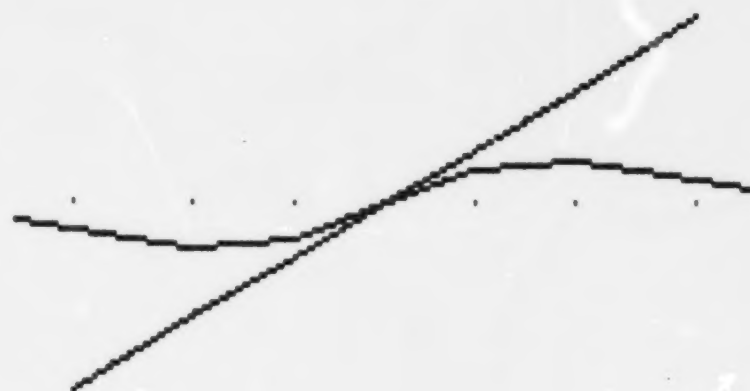
(4) Times included in the headings are times from initiation of braking.



HINGED6400 2T-IN-.5S DISP*10 L-STRUTS-ONLY 31AUG87
 TIME= 0.200 SECS. MAX DISPL= 5.189



HINGED6400 2T-IN-.5S DISP*10 L-STRUTS-ONLY 31AUG87
 TIME= 0.250 SECS. MAX DISPL= 10.075



HINGED6400 2T-IN-.5S DISP*10 L-STRTS-ONLY 31AUG87
TIME= 0.300 SECS. MAX DISPL= 17.335



HINGED6400 2T-IN-.5S DISP*10 L-STRTS-ONLY 31AUG87
TIME= 0.350 SECS. MAX DISPL= 27.194



HINGED6400 2T-IN-.5S DISP*10 L-STRTS-ONLY POST-IMPACT 9SEPT87
 TIME= 0.475 SECS. MAX DISPL= 8.691



HINGED6400 2T-IN-.5S DISP*10 L-STRTS-ONLY POST-IMPACT 9SEPT87
 TIME= 0.500 SECS. MAX DISPL= 12.984

APPENDIX D

Transient Behaviour of Indal 6400 with Hinges
under Braking Loads. Displacement Subsequent to
Strut-Strut Impact.

- NOTE (1) These displaced shapes, which are multiplied by a factor of 10.0, are plan views of the lower strut assembly only.
- (2) The dotted lines indicate the configuration at strut-strut impact.
- (3) The braking torque was increased linearly from zero to two times rated power torque (187,500 ft.lb., 256,000 N.m) in 0.5 seconds and thereafter remained constant.
- (4) Times included in the headings are times from initiation of braking. Strut-strut contact occurs at 0.350 seconds.



HINGED6400 2T-IN-.5S DISP*10 L-STRAITS-ONLY POST-IMPACT 9SEPT87
 TIME= 0.375 SECS. MAX DISPL= 4.322



HINGED6400 2T-IN-.5S DISP*10 L-STRAITS-ONLY POST-IMPACT 9SEPT87
 TIME= 0.400 SECS. MAX DISPL= 6.296



HINGED6400 2T-IN-.5S DISP*10 L-STRLTS-ONLY POST-IMPACT 9SEPT87
 TIME= 0.425 SECS. MAX DISPL= 8.190



HINGED6400 2T-IN-.5S DISP*10 L-STRLTS-ONLY POST-IMPACT 9SEPT87
 TIME= 0.450 SECS. MAX DISPL= 8.325